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On the Development of Optimally Efficient Compact Scroll Compressors for Refrigerators

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ABSTRACT

The present paper examines the development of a compact scroll compressor for refrigerators. Initially, the concept and geometry for a hybrid scroll compressor using a combination of an Archimedes' spiral and circular scroll are introduced. Subsequently, computer simulations undertaken to maximize the overall efficiency are presented for this hybrid scroll compressor. The optimal combination of major dimensions of the hybrid scroll compressor is identified and optimal design guidelines to maximize the overall efficiency are developed. Further, the possible maximum overall efficiency is estimated and shown to be comparable to that of a single involute scroll compressor. In addition, since the compressor is expected to operate under relatively high pressure ratios, the structural strength and deformation of the scroll raps must be carefully examined. To this end, FEM simulations were conducted for the hybrid scroll made of aluminum alloy to verify that it is sufficiently rigid for excessive overload operations.

1. INTRODUCTION

While scroll compressors have found application in heat pumps, their scaling is not well developed. One of the authors’ main concerns is the near-term enlargement and miniaturization of scroll compressors. In the present paper, the size reduction needed for a scroll compressor for use in residential refrigerators is considered. There is no other machine with such long-term and near continual use as the refrigerator. Therefore, refrigerators should have high efficiency and be quiet and non-obtrusive during operation. Scroll-type compressors with high efficiency and low noise characteristics may ultimately prove to be best compressors for refrigerators. This paper presents an academic research study on the development of a compact scroll compressor for refrigerators.

First, the hybrid scroll configuration is introduced. It consists of an outer Archimedes’ spiral smoothly merging with an inner circular profile, which makes it suitable for refrigerators operating under relatively high pressure ratios. Secondly, for given conditions of the scroll thickness and the leakage clearances, computer simulations of the mechanical, volumetric and compression efficiencies are undertaken for a variety of cylinder diameters, coefficients for the Archimedes’ spiral and scroll heights, to identify the optimal combination yielding the maximum overall efficiency. Finally, in order to verify that the hybrid scroll is sufficiently strong and rigid enough for overload opera-
2. CREATION OF ARCHIMEDES’ SPIRAL AND CIRCLE HYBRID SCROLL

An orbiting scroll consisting of two Archimedes’ spirals is presented in Figure 1(a), where the radius from the pole $O_m$ is represented by $f_i(\phi)$ for the inner scroll and by $f_o(\phi)$ for the outer, defined by

$$f_i(\phi) = a \cdot \phi + t,$$

in which $a$ is the spiral coefficient, $\phi$ the polar angle and $t$ the scroll thickness. When this cross-hatched scroll orbits around the fixed pole $O$ with radius $r_0$, many small circular trajectories are drawn, as shown in Figure 1(b) for the outer scroll and in Figure 1(c) for the inner. The outer envelope of the outer scroll trajectory circles in Figure 1(b) creates the inner fixed scroll, while the inner envelope of the inner scroll trajectory circles in Figure 1(c) creates the outer fixed scroll.

Figure 1 Creation of fixed scroll for a given orbiting scroll.

Figure 2 Configuration of Archimedes’ spiral compressor and replacement of the center portion spirals with circles.

In addition, FEM simulations were conducted to determine stresses and deformations of an aluminum alloy hybrid scroll, and showed that the hybrid scroll is indeed sufficiently strong and rigid enough even for excessive overload operations with CO$_2$ refrigerant.
As a result, an Archimedes’ spiral compressor can be created with the configuration shown in Figure 2(a), where the fixed scroll is not cross-hatched.

For a given orbiting scroll (cross-hatched in Figure 2(a)), the orbiting radius \( r_0 \) should be chosen as

\[
r_0 = \pi a-t,
\]

in order to create a fixed scroll with the same scroll wrap thickness \( t \). One difficulty is that the beginning of the fixed scroll is tapered. In order to overcome this difficulty, the center portions of the spirals are replaced with circles, as shown in Figure 2(b), in which the smaller circle with radius \( R_S \) forms the head of the scrolls, while the larger circle with radius \( R_L \) forms the inner curve of the scrolls. Points \( A \) to \( D \) are the intersections of the Archimedes’ spirals with the circles, where all corresponding spiral and circle tangents are required to match. When \( R_L \) is chosen to be

\[
R_L = r_0 + R_S,
\]

the smaller circles at the scroll heads always inscribe the corresponding larger circle. In addition, if the smaller and larger circles forming the orbiting or fixed scroll circumscribe, that is, if the line connecting the centers satisfies the following condition:

\[
\frac{O_S}{O_L} = R_S + R_L,
\]

the small circle can be connected smoothly to the large circle at Points \( E \) and \( F \). Adjusting the position of Point \( A \) along the Archimedes’ spiral, permits the fine adjustment of the radii \( R_S \) and \( R_L \).

A compression sequence of the hybrid scrolls is shown in Figure 3, where the two crescent-shaped chambers at the periphery of the scrolls are just closed at the orbiting angle \( \theta = 0^\circ \). The trapped chambers decrease in area to join just after passing \( \theta = 270^\circ \), when the center chamber area decreases to zero. Therefore, the compression chamber exhibits no jump in area, as shown in Figure 4(a). In addition, the decrease in area is much faster than for the involute scrolls, thus resulting in much faster increase in pressure, as shown in Figure 4(b). Thus, one may then expect that the Archimedes’ spiral and circle hybrid scroll will be substantially better for refrigerators operating under higher pressure ratios.
3. OPTIMAL DESIGN GUIDELINES FOR POSSIBLE MAXIMUM OVERALL EFFICIENCY

The scroll wrap profile is determined by the combination of the Archimedes’ spiral coefficient \(a\), the wrap height \(B\) and the cylinder diameter \(D\) in addition to the wrap thickness \(t\), for a given suction volume \(V_s\). Depending upon the
combination of these dimensions, the scroll wrap profile drastically changes, as does the gas pressure in the compression chambers, as shown in Figure 5, where $a$ is changed from 1.4 mm (on the left) to 2.8 mm (on the right) for fixed values of $D=67.54$ mm and $t=3.0$ mm. For an assumed suction volume of $V_s=5.0$ cm$^3$, $B$ becomes 11.5 mm for $a=1.4$ mm and 4.5 mm for $a=2.8$ mm. For $a=1.4$ mm, three different compression chambers co-exist simultaneously, while for $a=2.8$ mm, compression from the suction pressure $P_0=0.09$ MPa to the discharge $P_d=0.82$ MPa occurs in a single central chamber. Depending upon these drastic changes in gas pressure, there will be substantial differences in the refrigerant leakage through the wrap clearances, as well in the friction loss at the sliding pairs. The leakage, in turn, significantly affects the volumetric and compression efficiencies, while the friction loss significantly affects the mechanical efficiency, thus ultimately affecting the overall efficiency.

### 3.1 Volumetric Efficiency $\eta_v$

The volumetric efficiency $\eta_v$ can be calculated as the ratio of the actual discharge mass flow rate $G_e$ [kg/s] to the theoretical mass flow rate into compressor suction port, $G_{th}$ [kg/s]:

$$\eta_v \equiv \frac{G_e}{G_{th}} = \frac{G_{th} - G}{G_{th}}, \quad (5)$$

where $G_e$ is calculated by subtracting the leakage mass flow rate, $G$, from $G_{th}$. This leakage flow rate through the small clearances caused by the pressure difference between compression chambers in the scroll compressor can be determined using the simple theory by Ishii et al. (1996a) and Oku et al., (2005, 2006). The overall leakage mass flow rate for one revolution of the scroll’s orbiting movement can be calculated by integrating the local differential leakage mass flow rate over the length of the scroll wrap.

Once the scroll wrap profile is determined from the given parameters, such as compressor size, motor power and refrigeration capacity, the volume and the pressure in the compression chambers formed between the orbiting and fixed scrolls can be calculated geometrically. Subsequently, the leakage flow can be calculated.

To calculate the leakage flow through the small clearances in scroll compressors, the Darcy-Weisbach equation for incompressible and viscous fluid flow was applied, using a friction factor $\lambda$, shown in Figure 6 for CO$_2$. It is of

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significance to note that the corresponding empirically determined friction factor for R22 has been found to be almost indistinguishable from that for CO$_2$ (Oku et al., 2005). On this basis, it is assumed that the friction factor can be considered to be independent of the refrigerant itself. With this assumption, the friction factor $\lambda$ for R134a can be given as:

$$\lambda_a = 3.38 \text{Re}^{-0.46} \quad \text{and} \quad \lambda_r = 3.70 \text{Re}^{-0.46},$$

(6)

where $\lambda_a$ is the friction factor for the axial clearance and $\lambda_r$ is that for the radial clearance, for the surface roughness of 0.2 $\mu$m.

### 3.2 Mechanical Efficiency $\eta_m$

The mechanical friction (between the crankshaft and the crank journal, the crank pin and the orbiting scroll, the orbiting scroll and the Oldham ring, and the orbiting scroll and the thrust bearing) is the major source of power loss in scroll compressors. These mechanical friction loads can be obtained from a dynamic analysis for each pair of machine elements, and then summed to determine the overall friction force.

Dynamic equilibrium analysis yields the equation of motion governing the behavior of the crankshaft rotation given in the following expression (Ishii, et al., 1992):

$$\left( I_o + m_r r_o^2 + m_a r_a^2 \sin^2 \theta \right) \dot{\theta} + m_r r_o^2 \sin \theta \cos \theta \dot{\theta}^2 = N - \left( F_f + L_o + L_s + (f_{i1} + f_{i2}) \cdot r_o \sin \theta + (f_{i1} + f_{i2}) \cdot r_o \cos \theta \right)$$

(7)

where, the terms on left side represent the inertia torques and those on the right side represent the motor drive torque $N$ and the torque loads due to the gas compression, the mechanical friction torques at the crankshaft, the crankpin, the Oldham ring and the thrust slide-bearing. The mechanical friction forces are obtained as the product of the resultant force at each pair of compressor elements and the corresponding friction coefficient. The equation of motion (Eq. 7) can be solved numerically for the given torque characteristic of the electric motor and the pressure in the compression chambers to obtain a periodic solution.

Integrating the equation of motion of the crankshaft over the duration of one revolution of the crankshaft, an energy balance can be obtained. The shaft input power $W_s$ [N m/s], the gas compression power $W_i$ [N m/s] and the friction power loss $W_f$ [N m/s] are given by the following expressions, respectively:

$$W_s = n \int_0^{T_r} N \dot{\theta} dT \quad \text{and} \quad W_i = n \int_0^{T_r} F_f r_o \dot{\theta} dT$$

$$W_f = n \int_0^{T_r} \{ L_o + L_s + (f_{i1} + f_{i2}) r_o \} \dot{\theta} dT,$$

(8)

where $n$ is the rotational speed [rps]. The mechanical efficiency $\eta_m$ can then be calculated as:

$$\eta_m = \frac{W_i}{W_s} = \frac{W_s - W_f}{W_s}.$$

(9)

### 3.3 Compression Efficiency $\eta_c$

Due to the continual re-compression of the leakage flows and the dissipation of energy through the frictional losses associated with the leakage flow, a scroll compressor with leakage requires greater compression power than one...
with no leakage. Therefore, the compression efficiency $\eta_c$ can be defined by the ratio of the theoretical gas compression power with no leakage, $E_{th}$ [Nm/kg], to the real gas compression power with leakage, $E$ [Nm/kg]:

$$\eta_c \equiv \frac{E_{th}}{E}, \quad (10)$$

which takes on a value that is then less than 1.0.

### 3.4 Resultant Efficiency $\eta$

The overall efficiency $\eta$ can be obtained as the product of the component efficiencies $\eta_v$, $\eta_m$ and $\eta_c$:

$$\eta = \eta_v \times \eta_m \times \eta_c \quad (11)$$

which represents the ratio of the accumulated energy in the discharged refrigerant to the shaft input power $W_p$.

### Table 1 Major specifications of an Archimedes’ spiral and circle hybrid scroll compressor for refrigerators, for computer simulations of performance.

<table>
<thead>
<tr>
<th>Operation conditions</th>
<th>Suction volume $V_s$ [cc]</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adiabatic index $k$ [-]</td>
<td>1.12</td>
<td></td>
</tr>
<tr>
<td>Pressure ratio $p_r$ [-]</td>
<td>9.11</td>
<td></td>
</tr>
<tr>
<td>Compression ratio $p_c$ [-]</td>
<td>7.19</td>
<td></td>
</tr>
<tr>
<td>Operation speed $f_{rev}$ [rpm]</td>
<td>3498</td>
<td></td>
</tr>
<tr>
<td>Mechanical constants</td>
<td>Radius of Crank jounal $r_o$ [mm]</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Rotation radius of Crank gravity $r_g$ [mm]</td>
<td>1.01</td>
</tr>
<tr>
<td></td>
<td>Mass of Oldham's ring $m_O$ [kg]</td>
<td>0.037</td>
</tr>
<tr>
<td></td>
<td>Inertia moment of Crank shaft $I_o$ [kgm²]</td>
<td>0.0011</td>
</tr>
<tr>
<td></td>
<td>Radius of Crank-pin $r_p$ [mm]</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Mass of Crank-pin $m_{CP}$ [kg]</td>
<td>0.0054</td>
</tr>
<tr>
<td></td>
<td>Radius of Ball bearing $r_{ball}$ [mm]</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Fric. Coef. Of Oldham's ring $\mu_o$ [-]</td>
<td>0.055</td>
</tr>
<tr>
<td></td>
<td>Fric. Coef. Of thrust bearing $\mu_t$ [-]</td>
<td>0.055</td>
</tr>
<tr>
<td></td>
<td>Fric. Coef. Of Crank pin $\mu_p$ [-]</td>
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</tr>
<tr>
<td></td>
<td>Fric. Coef. Of Crank bearing $\mu_\theta$ [-]</td>
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</tr>
<tr>
<td></td>
<td>Fric. Coef. Of Crank ball bearing $\mu_{ball}$ [-]</td>
<td>0.0013</td>
</tr>
</tbody>
</table>

### 3.5 Calculated Results of Efficiencies

The major specifications for computer calculations are shown in Table 1, where the operation conditions and the major mechanical constants of the compressor are specified. The refrigerant is R134a for refrigerators, where the condensation temperature is 30°C at the discharge pressure of 0.82 MPa, and the evaporation temperature is -30°C at the suction pressure of 0.09 MPa. The resulting pressure ratio is 9.11. The suction volume $V_s$ is fixed at 5.0 cm³, assuming a small cooling capacity compressor for refrigerators. Based on FEM examinations for structural strength of the scroll wrap (see the following section for details), the wrap thickness $t$ is fixed at 3.0 mm. As a result, for given values of the cylinder diameter $D$ from 50 mm to 90 mm and the Archimedes’ spiral coefficient $a$ from 1.4 mm to 3.0 mm, the suction area $S_s$, trapped at the periphery of the scrolls, can be calculated, as shown in Figure 7(a), and subsequently the scroll wrap height $B$ can be calculated by $V_s/S_s$, as shown in Figure 7(b). Representative points A and B in this figure yield the scroll configurations, shown in Figures 5(a) and 5(b), respectively.

With decreased suction volumes found in refrigerators, the leakage clearances have to be as small as possible to produce high volumetric efficiency. From this consideration, an axial clearance of $\delta_a=3$ μm and a radial clearance of $\delta_r=6$ μm are assumed in the present development. The friction coefficients, as is usual, are taken to be 0.055 for the thrust slide bearing, 0.011 for the crank pin and crank journal and 0.0013 for the ball bearing at the end of the crank shaft, all which were measured by friction tests on small cooling capacity compressors.
Leakage, friction and pressure-volume relationships will affect efficiency. Figures 8 and 9 show examples of how these parameters vary with the spiral coefficient $a$. First, the leakage flow velocity and leakage mass were calculated for both the axial and radial clearances. The leakage mass $\Delta G$ for one cycle of orbiting motion is shown in Figure 8(a), in which the abscissa is the spiral coefficient $a$. As $a$ increases, the pressure difference between the suction chamber and the inner compressed chamber increases, as shown in Figure 5, resulting in an increase in the refrigerant gas leakages through the radial and axial clearances, $\Delta G_r$. 

Figure 7  Suction area and wrap height vs. Archimedes’ spiral coefficient, for $t=3.0\text{mm}$ and $V_s=5.0\text{cc}$

Figure 8 Leakage and friction loss vs. spiral coefficient.

Figure 9 Ideal and real Pressures vs. volume.

Figure 10 Calculated efficiencies and speed variation ratio vs. spiral coefficient, for $D=67.54\text{ mm}$.
and $\Delta G_a$. The resultant total refrigerant gas leakage $\Delta G$ increases from $1.5 \times 10^{-6}$ kg to $3.5 \times 10^{-6}$ kg as $a$ increases from 1.4 mm to 2.8 mm and hence the volumetric efficiency $\eta_v$ decreases from its maximum value of 91.8 % at $a = 1.4$ mm to 80.5 % at $a = 2.8$ mm, as shown in Figure 10(a).

Figure 8(b) shows the calculated friction power losses. As the spiral coefficient $a$ increases, the friction loss $W_{v,p}$ at the crankpin and $W_{v,s}$ at the crankshaft decrease gradually, since the wrap height decreases and, hence, the gas loads on the crankpin and crankshaft decrease. On the contrary, the friction loss $W_{t,b}$ at the thrust slide-bearing increases with increasing $a$, because the chamber bottom area increases and, hence, the gas thrust force on the thrust slide-bearing increases. The resultant friction loss $W_f$ exhibits a convex trend, initially decreasing to 10.4 W at $a = 1.83$ mm and then to increasing. From these results, the mechanical efficiency $\eta_m$ can be obtained as shown in Figure 10(b), in which the mechanical efficiency $\eta_m$ exhibits its maximum value of 88.6 % at $a = 1.83$ mm. The crankshaft speed fluctuation ratio $D$ increases from 0.83 % to 3.15 %, with increasing $a$ from 1.4 mm to 2.8 mm.

The $P-V$ diagrams for $a = 1.4$ mm and 2.8 mm are shown in Figures 9(a) and (b), respectively, in which the theoretical gas pressure $P_T$ is plotted by the dashed line and the actual gas pressure $P_R$ by the solid line. The value of the compression efficiency is governed predominantly by the in-flow leakage to the outer compression chamber from the higher pressure compression chamber, which has a far larger effect than the leakage out-flow to the suction chamber. As a result, for the small $a$ range, shown in Figure 9(a), $P_R$ becomes far larger than $P_T$, thus resulting in lower compression efficiency. As $a$ increases, on the contrary, the in-flow leakage to the compression chamber decreases, and then $P_R$ approaches to or decreases below $P_T$ curve, as shown in Figure 9(b). Correspondingly, the compression efficiency $\eta_v$ increases from 75.9 % at $a = 1.4$ mm to 102.8 % at $a = 2.8$ mm, as shown in Figure 10(d).

Finally, the overall efficiency $\eta$ can be calculated by Eq. (7), as shown in Figure 10(e), where the three efficiency components are multiplied. While the mechanical efficiency $\eta_m$ exhibits its maximum at the spiral coefficient $a = 1.83$ mm, the overall efficiency $\eta$ exhibits its maximum value of 68.9 % at $a = 2.1$ mm, predominantly affected by the rapid increase in the volumetric efficiency.

Similar calculations were conducted for a variety of the cylinder diameter $D$ from 50 mm to 90 mm, in order to determine the spiral coefficient $a$, at which the overall efficiency exhibits its maximum value $\eta_{max}$. Calculated results of $\eta_{max}$ are shown in Figure 11(c), where the maximum overall efficiency increases with increasing the cylinder diameter.
diameter $D$ and reaches its maximum of 68.7% at $D=67.54$ mm. On this basis, the optimal design guideline for the Archimedes’ hybrid scroll is 67.54 mm for the cylinder diameter, $a=2.1$ mm for the spiral coefficient shown in Figure 11(a) and $B=5.2$ mm for the wrap height shown in Figure 11(b). Under these conditions, the crankshaft speed fluctuation ratio $\alpha$ is 1.81%, as shown in Figure 11(d). The scroll wrap profile of Archimedes’ hybrid compressor suggested by these guidelines is shown in Figure 11(e), where the aspect ratio ($=B/D$) of the cross-section is 0.078.

4. FEM SIMULATION OF STRUCTURAL STRENGTH OF OPTIMIZED SCROLL WRAP

Structural strength of the optimized scroll wrap for refrigerators, shown in Figure 11, was carefully examined with FEM simulations for stress and deformation. PRO/MECHANICA2000i P-Method with higher order deformation functions was used for the present FEM simulations, where simulated results with higher accuracy can be obtained with relatively few element divisions, the number of which was fixed at 20. Material of the scroll wrap was assumed made of aluminum alloy with a creep stress of 415 MPa. With the assumption that the present scroll wrap might well be used as CO$_2$ compressor, the suction pressure was fixed at 3.5 MPa, while the discharge pressure was increased to 9 MPa.

FEM-simulation results are shown in Figure 12, in which the equivalent single involute scroll also presented for comparison. The abscissa is the overload ratio relative to the base discharge pressure of 9 MPa. The colored fringe display for the stress is a view from the thrust disk, while that for the deformation is from the scroll wrap tip. For the Archimedes’ hybrid, the position of maximum stress and deformation is about at the half circle away from the beginning, while for the single involute, the position arises near the beginning. The Archimedes’ hybrid scroll exhibits comparatively small values both in the stress and deformation. Even for 100% overload, the stress is 156 MPa that is far smaller than the safety factor of 2.0, and the deformation is 4.5 μm which is sufficiently smaller than the assumed radial clearance of $\delta_r=6$ μm.

![FEM simulations of stress and deformation of optimized scroll wrap](image)

Figure 12 FEM simulations of stress and deformation of optimized scroll wrap.

5. CONCLUSION

Efficiency calculations of the Archimedes’ spiral-and-circle hybrid scroll compressor for refrigerators operated with refrigerant R134a were undertaken to determine the optimal design values of Archimedes’ spiral coefficient, scroll
height and cylinder diameter for a fixed suction volume of 5.0 cm\(^3\) and scroll thickness of 3.0 mm. The assumed radial clearance between the orbiting to fixed scrolls was 6 \(\mu\)m and the assumed axial clearance was 3 \(\mu\)m. The temperature and pressure were fixed at -30°C and 0.09 MPa for the evaporation and 30°C and 0.82 MPa for the condensation. The R134a leakage flow through axial and radial clearances between the fixed and orbiting scrolls was calculated with application of the friction factor for leakage of CO\(_2\) and R22 refrigerants to obtain the volumetric and compression efficiencies, whereas the friction losses were calculated to determine the mechanical efficiency. As a result, the maximum overall efficiency of 68.7 % was found for the Archimedes’ spiral coefficient of 2.1 mm, scroll height of 5.2 mm and cylinder diameter of 67.54 mm. Finally, it was confirmed with FEM simulations that the optimized scroll wrap is sufficiently rigid even for use with CO\(_2\) refrigerant.

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